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EXPOSURE

vol.6 no.1 a newsletter for ocean technologists

A Velo~~s~~city-Sensitive, Bottom-Contact Release

A bottom-contact release was needed for a deep ocean data acquisition system which was required to freefall to a depth of 6100 m and return to the surface for recovery. The system would be housed in a 19 cm OD cylinder containing the data acquisition circuits, pressure transducer, and transponder electronics (which allows the system to be tracked *in situ*). A connector on the bottom end cap would provide electronic access to a variety of instruments mounted on the outside of the cylinder.

The bottom-contact release design specifications required that it interface with a parallel acoustic command system which could be used to terminate the sounding before bottom contact or as a backup release. The unit was also required to operate with any type of bottom condition and use an inexpensive, easily replaced, anchor. Nominal descent and ascent velocity was set at 1.8 m/sec, giving a round trip time of 1.8 hours with the 6100 m maximum operating depth of the system. A purely mechanical system was desired to insure proper operation in the event of power failure. Thus, electromagnetic devices and bottom-contact switches were eliminated. Timing devices, pressure switches, and dissolving links were also ruled out because we wanted the system to drop its ballast immediately upon reaching the bottom. Other release methods using counterweighted mechanisms were considered undesirable because of possible premature release during launch.

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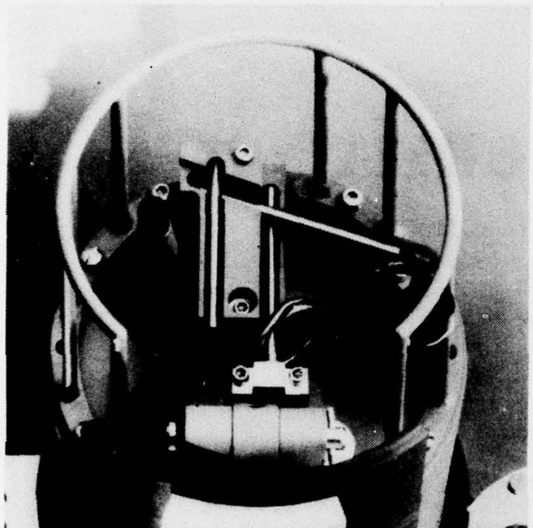
Considering the initial requirements, we decided a velocity-sensitive release would be best. The present design, using this concept, is shown in Figure 1. There are three pins placed in channels on the end of the cylinder. These pins are placed one above the other in such a way that until the descent velocity dropped below 30 cm/sec. To prevent release during launch the vane is initially held in place by a pressure release pin. The pin, as shown in Figure 2, is attached to a small piston mounted in its own pressure housing. At a depth of approximately 30 m,



VELOSITY-SENSITIVE RELEASE

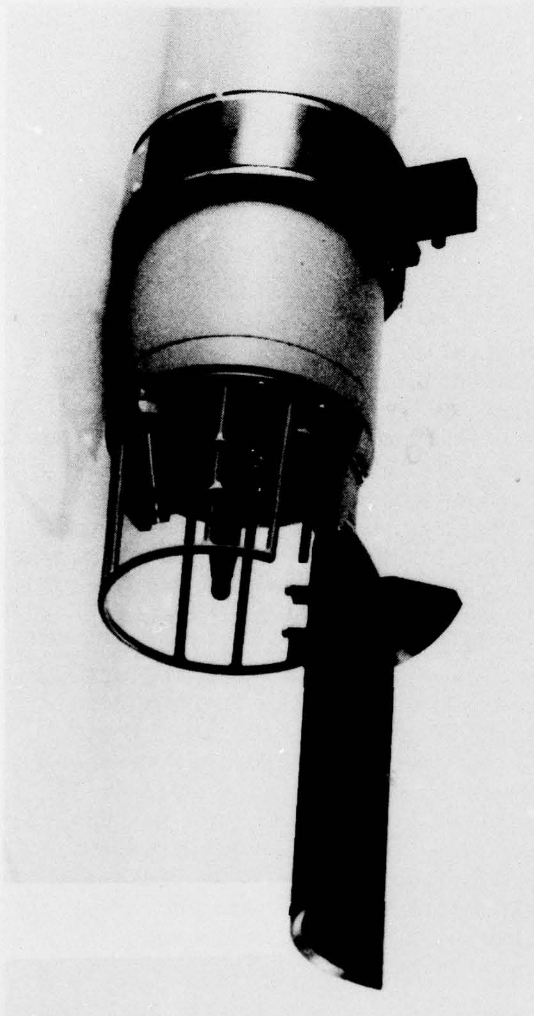
Figure 1.

THE VANE HELD IN PLACE BY THE PRESSURE RELEASE PIN

Figure 2.

freeing one end of either of the uppermost two pins allows the lowest pin to drop, thus releasing an oval link to which the anchor line is attached. One end of the upper pin is held in place by an electrolytic release wire, while the other end is retained by a cam which is part of a pivoted vane assembly used as the velocity-sensing device. The vane pivot is secured to the end cap, and the vane itself projects from the pivot along the pressure housing. As long as the system is descending, the water flowing by the housing holds the vane in place. Sea tests showed that the pin was not released

the pin is retracted sufficiently to free the vane. Once the pressure-release pin is retracted, the release is dependent upon the water-flow pressure on the vane to maintain the vane in a restraining position. When the descent velocity decreases, as when the anchor hits the bottom, the vane pivots away from the housing and pulls the cam away from the bottom-release pin. The pin drops and frees the end of the electrolytic release pin over which it was installed. The electrolytic release pin, now retained only by the release wire at its other end, pivots out of the way of the anchor retaining pin, which drops out of the end cap, thus freeing the steel oval link and anchor line. As shown in Figure 3, the vane was designed to swing more than 180° after release so that it would present minimum drag during ascent.

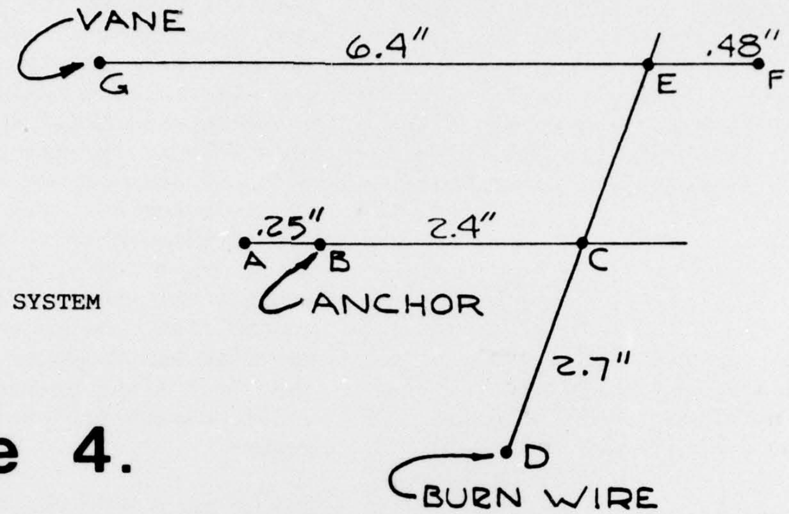


THE VANE IN ASCENT POSITION
WITH RELEASE PINS STILL IN PLACE.

Figure 3.

The primary, and interrelated, forces influencing the design of the bottom release are anchor weight and drag due to the descent rate. Equally important secondary forces in the system design are drag on the vane, friction on the vane's hinge and cam, and the weight of the vane. To use the system with a wide range of anchor sizes, it was necessary to reduce the effect of anchor weight on the velocity-sensing vane. To accomplish this, a three-pin leverage system, as shown in Figure 4, was devised which reduced the anchor force at the vane to .99 percent of the total wet weight of the anchor. For example, if a 453 kg anchor is attached at point A, the vane feels 2.2 kg at point G. The anchor force is so reduced at the vane that the assembly is found to be limited by the bending stress in the release pin, and the 29.4 kg breaking strength of the release wire, and not by the anchor weight force at the vane. Tests show the release pin realizes permanent deformation at anchor loads of approximately 294 kg.

The bottom-contact release system is also required to function over a wide range of descent velocities. Small descent velocities are critical in this case. If the



RELEASE PIN SYSTEM

Figure 4.

descent velocity of the system is too small, the drag force on the vane will cause it to prematurely rotate downward and release the anchor. One solution is to make the vane very large to get the increased drag, but that approach has obvious limitations. The best approach was to decrease the sensitivity of the vane. This was done by allowing the vane to rotate a large angle before releasing the pins. It should be obvious that as the projected area of the vane increases due to the vane's rotation, the drag force also increases. The increased drag force on the vane allows for lower descent velocities at release. A 60° critical angle of release was chosen. A 90° critical release angle could be chosen to further decrease the sensitivity of the vane so that a wider range of descent velocities could be handled by the system. With the vane's 60° release angle, the system will release at a descent velocity between 15 and 30 cm/sec.

The vane, approximately 26 by 14.7 cm, is constructed from a section of PVC tubing, and it is nearly neutrally buoyant in seawater. Counterweights near the base of the vane were attached to act against the frictional forces on the hinge and to allow the vane to rotate downward upon bottom contact. The counterweights are zinc and weigh 227 gm each. The zinc counterweights also serve as anodes to protect the aluminum release parts against galvanic corrosion. The counterweights, shown in Figure 2, are placed close to the hinge so that the moment about the hinge due to the counterweights remains almost constant during the vane's rotation. This was done so that the drag force and counterweight moments about the hinge would not cancel each other near the critical release angle at low descent velocities.

The force of the release pin on the vane for any vane rotation causes friction on the cam surface and on the hinge pin. This means a moment is present which resists vane rotation because the release pin is designed so that its end is vertically in line with the center line of the hinge pin and no direct

righting or upsetting moment is present about the hinge. As was previously stated, the anchor force has little effect at the vane and is as small as possible to keep friction on the vane's cam and hinge minimized. Other steps taken to reduce the dampening effect of frictional forces on the system were the use of teflon bushings around the vane pin, the machining of the cam's surface to a 64 RMS finish, and the chamfering of the end of the release pin. Frictional forces were minimized and therefore the dampening effect of frictional righting or upsetting moment was reduced to a point that it has little effect on the system's performance--even with the large anchor loads.

The acoustic-release mechanism, an electromechanical assembly, is the backup release and it will function in the case of a mechanical failure of the velocity-sensing system. The electromechanical system consists of a burn wire, release pin, and electronics. The burn wire is an Inconel® 625 wire which has a breaking strength of approximately 45.3 kg and maintains very good corrosion properties in seawater. It was found that the load at the release wire is .024 percent of the anchor load. For a 453 kg load, a force of 118 gm is felt at the release wire; this is well within the load carrying capability of the Inconel® wire. The release pin is a two part assembly: the section to which the release wire is terminated is a fiberglass section; and the remainder of the pin is mild steel. The two sections are press-fit together. The nonconductive fiberglass section electrically isolates the Inconel® wire from the rest of the assembly so when a

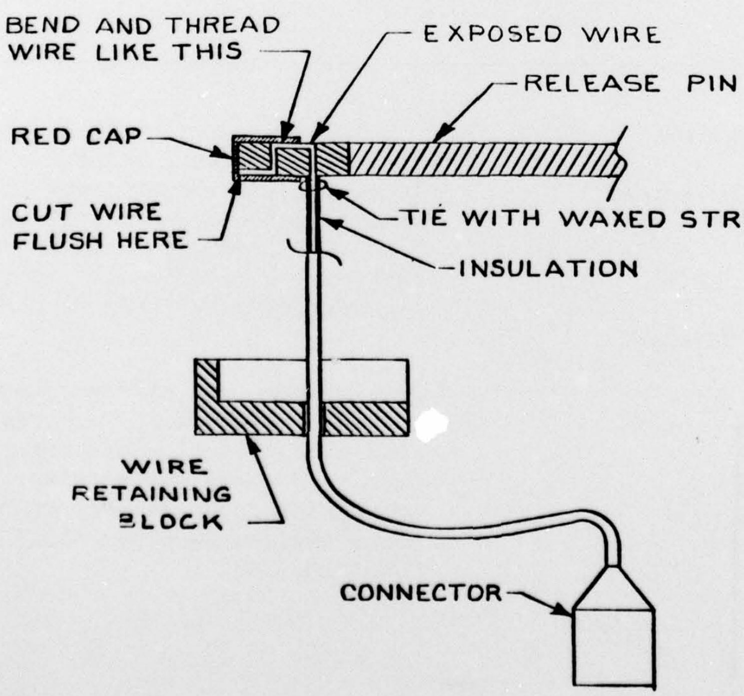


FIGURE 5.

positive voltage is applied to the wire in seawater, the Inconel® wire will plate away until it breaks. With a 2.46 kg anchor, release occurs in 7 to 10 minutes. Figure 5 shows the assembly of the release-wire system. Note the bends in the wire which cause the wire to bind and hold down the pin. The parallel acoustic-release system has been extensively tested and is known to perform in systems for a 5 year life span.

With the principles presented in the preceding paragraphs, a bottom-contact release could be designed for equipment with wide ranges of anchor weights and descent rates. This could be done by scaling the system up or down to meet the required specifications. The present system has been sea tested with a 59 kg anchor which yielded a 1.8 m/sec descent rate of the unit. Results show the system performed as expected with good results.

The preceding design is proprietary to Sonatech, Inc., Goleta, California.

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A Method For Faired-Cable WINCH Assembly

A 1.3 m diameter winch drum was needed to hold and deploy the 200 m of faired conducting cable used with the Distributed Instrumentation Profiling System (DIPS) described in the *Exposure* newsletter, Vol. 5, No. 3. Because they reflect the character of their application, most winches are custom designed and, therefore, become an expensive system component in time and dollars. For these reasons our winch acquisition problems were no different. The following article is a picture story illustrating a basic winch design, and a simplistic method of fabricating it with minimal tooling, that can provide the developer with greater time flexibility and cost savings.

Our requirements were for a unit that could hoist 700 kg at a line speed of 15 or 30 cm/sec. For less weight and surface maintenance, 6061-T6 aluminum stock was selected for the majority of metal parts.

The powered disc and outer support disc for the drum were cut round with a metal band saw, as shown in Figure 1. A circular tolerance of ± 1.6 mm was attainable with this method. Each disc took about 15 minutes to cut out once the initial set-up was made. Next, in Figure 2, the discs were gang drilled for all the hole patterns. In the disc leaning against the tool board, 24 holes are shown which serve as sockets for the spacer rods that

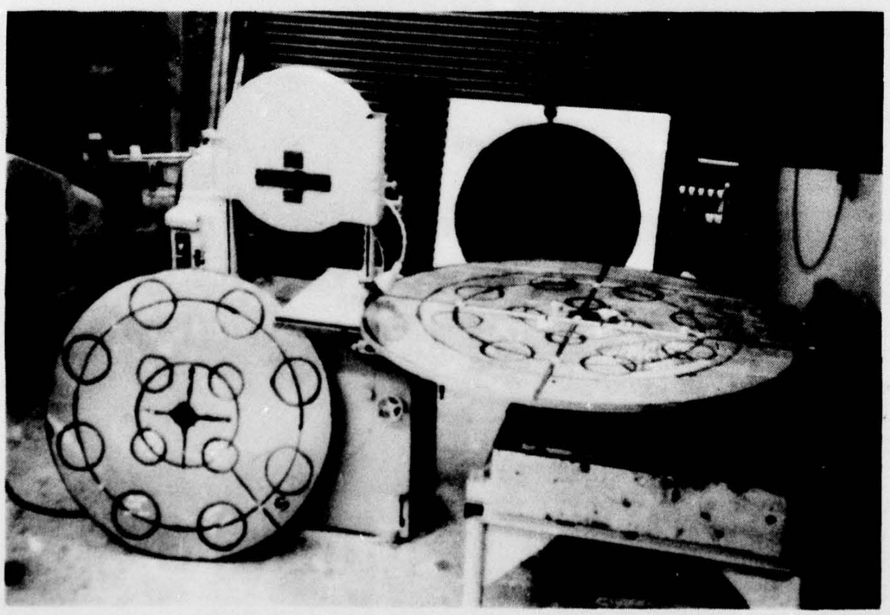


Figure 1.

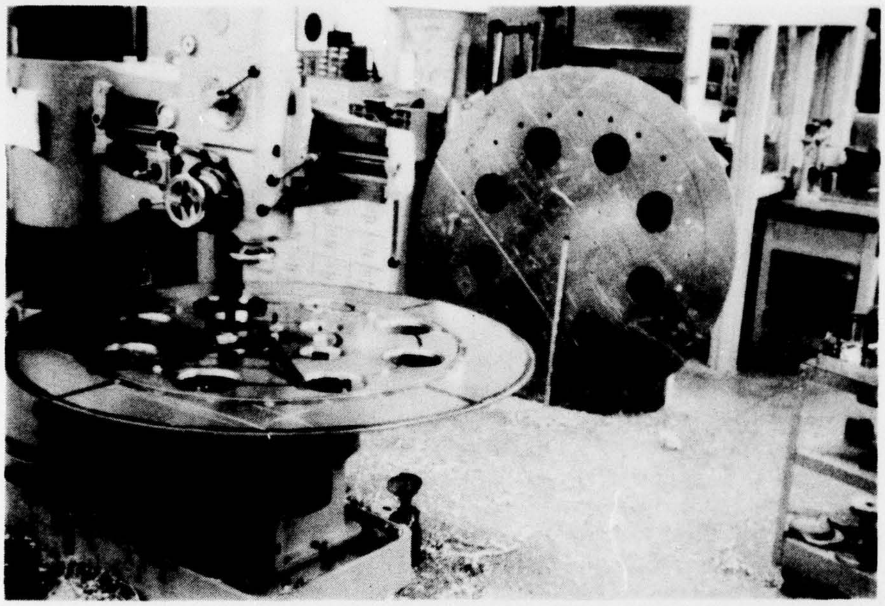


Figure 2.

form the drum support structure. Once the discs are clamped against each other with the spacer rods in place, as in Figure 3, a metal inert gas (MIG) welder is used to weld the outermost ends of the rods to the discs. The welding is minimal and fast on each rod end so that the heat build-up does not contribute thermal warping to the drum cage. In Figure 4, a 70 mm thick, prerolled skin is clamped over the support rods and the seam is welded together and ground flat. Additional 10 cm welds between the support rods are made from the inside to bond the band and the powered disc together. With this band in place the drum has all of its designed rigidity.

The winch base is shown in the forefront of Figure 4. This base is made from 12.5 cm structural "I" beam stock that has been cut on the band saw and welded into a box frame. The drum rotates on a 7.6 cm OD by 1.27-cm-thick wall steel shaft which is supported by pillow blocks that bolt to the base frame. The shaft is keyed to the drum, but it is not the drive attachment for the power

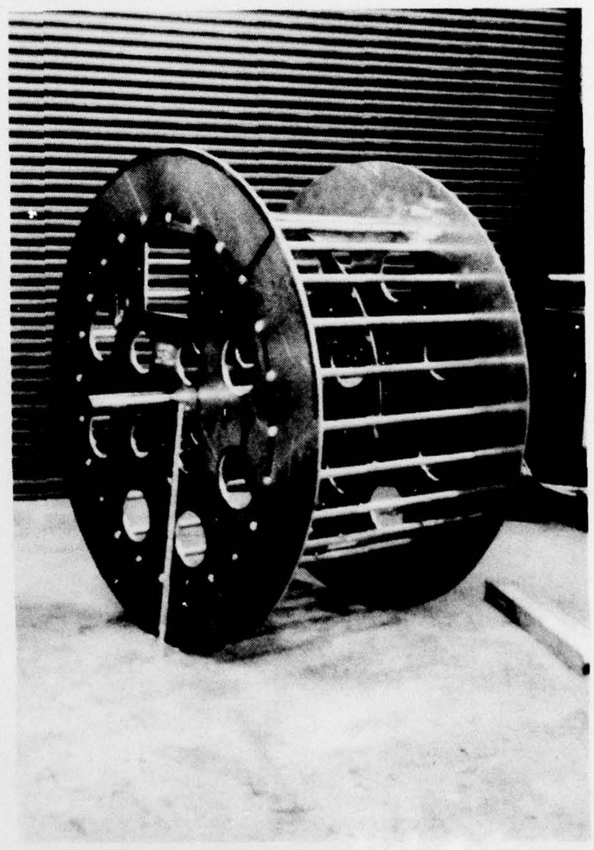
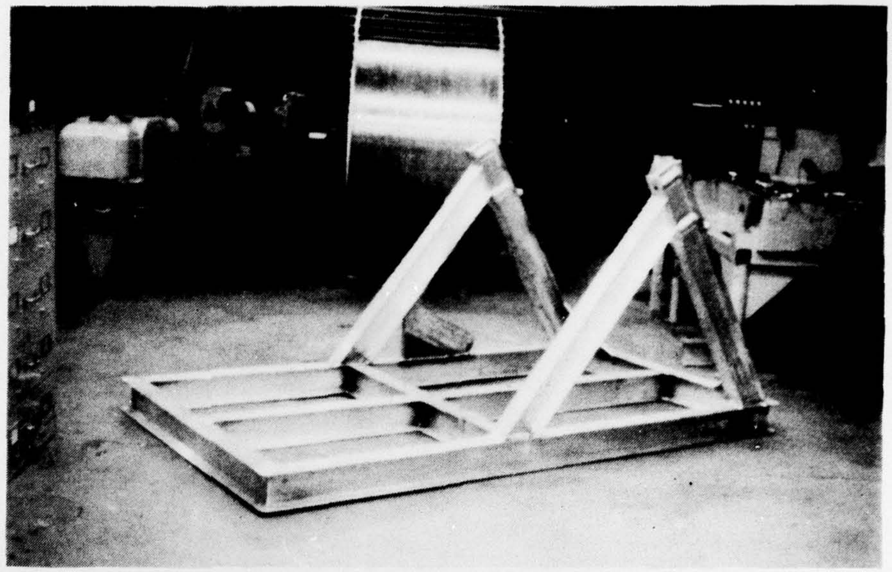


Figure 3.

Figure 4.



train. The power train shown in Figure 5 consists of a:

Baldor model M3710T motor, 220/440v/3Ø, with 213T frame, 1720 rpm, 7.5 hp.

The motor is coupled by double V-belt pulleys for a 2:1 speed reduction to an:

Ohio Gear Inc., gear train (1), model DHH-2-MC-184C-b3,

which provides a 40:1 speed reduction. This unit is directly coupled to a:

Ford Co. Model 14-17 gear train (2) hay baler gear box SN 2519,

which provides an 8:1 speed reduction.

A 28-tooth sprocket on the output shaft of gear train (2) coupled by number 60 Morse chain, to an 80-tooth

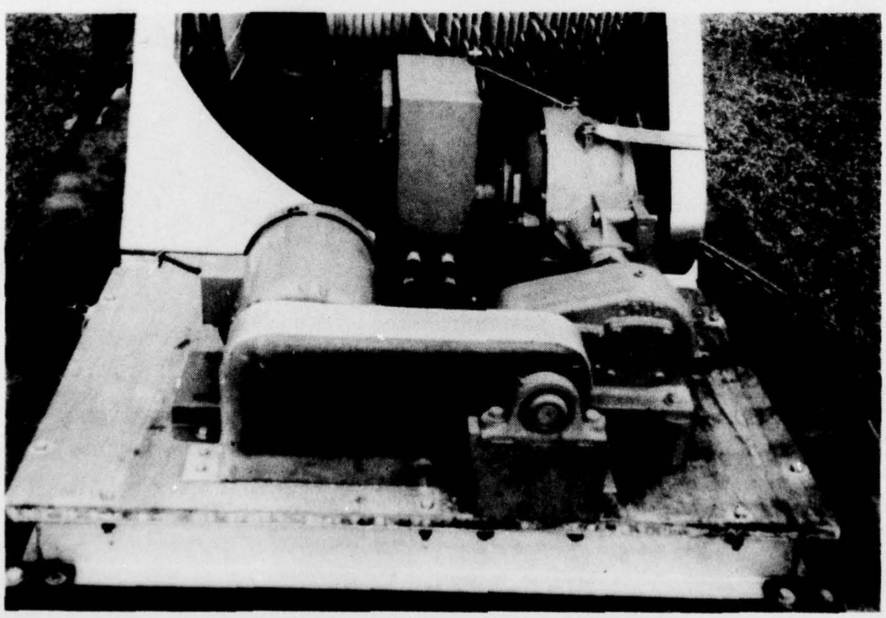


Figure 5.

sprocket attached to the disc on the drum, completes the 855:1 speed reduction necessary to provide the 15 cm/sec line speed.

The motor has a hand-held operator station with "activate," "forward," and "reverse" control buttons.

The power train is clustered on a 3/4-inch-thick aluminum plate that is bolted to the base frame. Both the power train and the drum can be easily unbolted from the base frame as unit assemblies.

Hydraulic automobile disc brakes, operating at the outer periphery on one of the drum discs, provide the drum breaking function. Two disc brake assemblies from a 1970-75 VOLVO automobile and a master cylinder from a similar year FORD automobile are securely attached to the base frame, as shown in Figure 6. The brake control, which includes an electrical cutout to the motor, can be manually operated from either side of the winch.

To prevent losing the conducting cable, in the event the drive chain parted, a mechanical drum lockup was incorporated into the winch design with the assembly shown in Figure 7. If the chain-idler-pulley arm is allowed to rotate 90 degrees due to a slack drive chain, a section of tool steel is driven into the edge of the drum disc to immediately block its movement.

Figure 8 shows the complete winch as it was mounted for a recent cruise.

WINCH STATISTICS

Base: 244 cm long, 129.5 cm wide, 12.7 cm high. The I-beam structural aluminum is 12.7 cm high with a 1.25 cm web.

Drum: Outer discs: 178 cm diameter, 1.27 cm thick; center disc: 130 cm diameter, 0.95 cm thick.

Support rods: 2.92 cm diameter, 94 cm long.

Approximate weight: 1000 kilograms

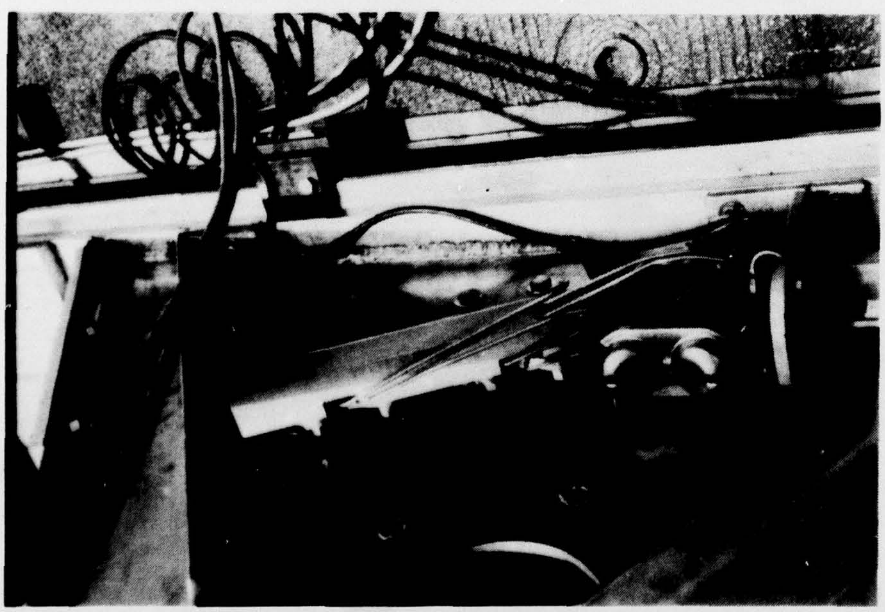


Figure 6.

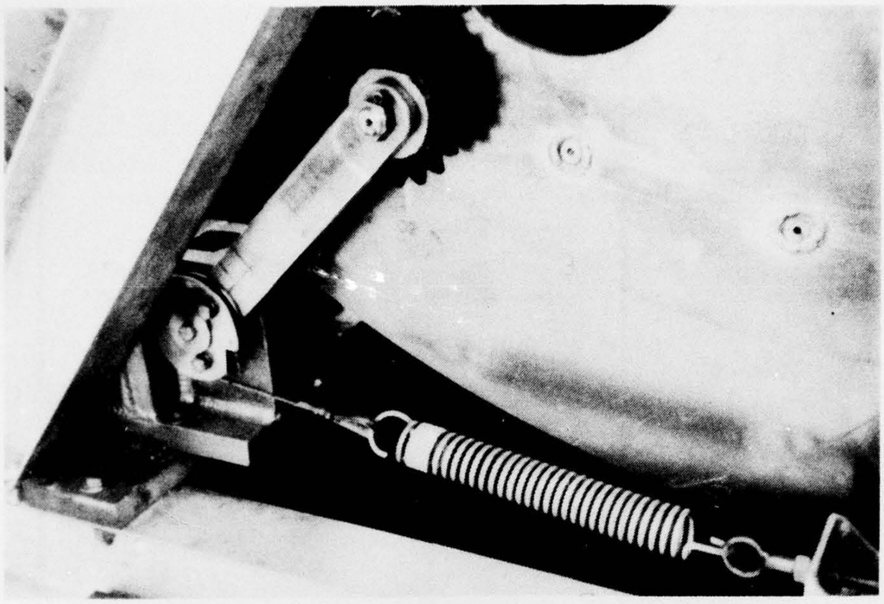
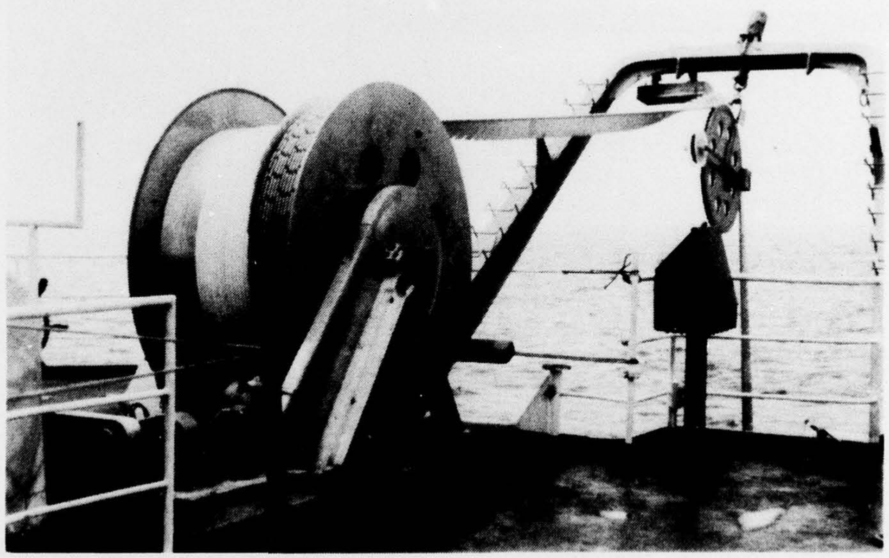


Figure 7.

Figure 8.



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OVER-->

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T A B L E   O F   C O N T E N T S

ANNOUNCEMENT:

Equipment users groups can be very helpful in the exchange of information on technical problems and data analysis techniques. For these groups to be effective, they need your support and interaction.

If you are interested in participating in an Aanderaa Current Meter Users group, please contact:

Sam D. Baird
Ocean Instrumentation/CCIW
P.O. Box 5050
Burlington, Ontario
Canada L7R 4A6
(416) 637-4709

Anyone similarly interested in participating in a users group for Endeco 174 current meters should contact:

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